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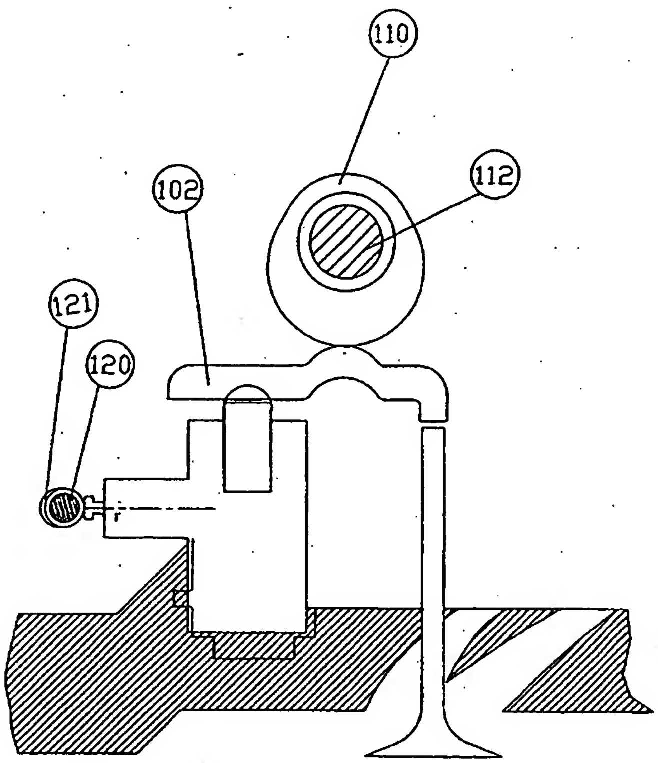
(54) Title: VARIABLE VALVE TIMING SYSTEM

(57) Abstract

A valve control arrangement for internal combustion engines in which a hydraulic valve actuator with integral fluid reservoir is interactive with two camshafts (110, 120) to continuously and independently vary the valve duration. A modified intake system is provided to increase the effective area of the valves.



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VARIABLE VALVE TIMING SYSTEM

BACKGROUND OF THE INVENTION

Field of the invention

This invention relates to the art of variable valve control system for internal combustion engines and more particularly, to a fully flexible valve timing system enabling control of load and improvement in effective valve area.

Description of the prior art

Strict emissions standards, coupled with requirements for improved fuel economy have mandated significant changes in the way automobile engines are designed.

Until recently, automotive engines with fixed valve timing have been the norm. As a result unless run at speeds near their design points, overall performance is less than ideal, resulting among others in the problem of backflow and charge dilution at low speed operating conditions.

Several solutions have been proposed to address this problem. These have mostly relied on fixed-lift, 2-position phasing. This consists mainly of a mechanism to rotate the intake camshaft with respect to the crankshaft to provide decreased overlap at low speed and late intake valve closing at high speeds. Since the valve lift profiles are fixed in terms of crank angle any variation in the closing angle tends to compromise the opening angle and vice versa.

A more sophisticated approach which minimizes this tradeoff has been to employ flexible electro-hydraulic actuators acting in response to timed signals from an ECU to modify the lift curve. Although, resulting in improved

performance over a greater load range, these electro-hydraulic systems do not fully overcome the problem of scavenging the residual exhaust gases arising as a result of the pumping losses sustained in the load control mechanism.

A fully flexible valve control system offers the optimal solution by eliminating pumping losses through the provision of the load control function. The above systems have not demonstrated the required flexibility to fulfill this objective except over a very narrow load range. Furthermore, due to the inherent problems of flow and time lag most of the current electro-hydraulic systems suffer from degradation at higher rpm. The objective is to design a variable valve actuation (VVA) mechanism with the capability to control load over the entire operating range. Recent approaches to achieving such an optimum control system have focused mainly on high speed solenoids or hydro-pneumatic methods. However, these schemes are regarded as complex and expensive and can impose a significant power drain on the engine.

In another approach a valve is operated by two camshafts through a system of levers. One such design is set forth in U.S. Patent, Serial No. 4,714,057. The practice taught by the patent limits the scope of the design. The limited phasing is not sufficient for load control. Any such phasing system would be subject to the considerable back pressure of the valve springs, needing some form of stabilization. Furthermore, the cost and space requirements are considerable.

Other designs for hydraulic control of the valves have been proposed in U.S. Pats. 4,615,306, 4,615,307 and 4,889,084, but these varyingly suffer from slow response time, degradation at high speed or high cost/space requirements.

Thus there has long been a need for a satisfactory arrangement to flexibly control the timing and duration of an engine's valves.

It is also desired to enable the valves to function as the optimal load control device.

It is also desired to maximize the air flow rate to the cylinders at all speeds and load.

It is yet another desire to maximize the effective area of the valves.

SUMMARY OF THE INVENTION

Accordingly, the present invention provides an improved valve opening and duration control arrangement.

It is yet another objective to improve valve actuation and sequencing flexibility to enhance performance and economy and reduce emissions.

It is yet another objective to eliminate pumping losses by enabling valves to control engine load.

It is yet another objective to double the effective area of the valves to improve engine breathing.

It is also an objective to achieve these objectives at a relatively low cost and with a high degree of reliability.

The above and other objectives of the present invention are achieved, according to a preferred embodiment thereof, by providing an improved control of valve open duration by progressively advancing the intake valve closing phase angle. The closing phase angle is shifted in a stepless manner. The arrangement functions without effecting the opening phase angle and overlap period and is equally adaptable for partial or full phasing of any of the valve events.

It is also observed that the backflow problem is avoided if the intake tubes are at or above ambient pressure at all times. Under such conditions the need for variation in the overlap region diminishes, obviating the requirement for phasing in this portion of the operational curve. This yields a simpler, more compact design with lower costs. The charge admitted to the cylinders can be a function of the valve duration in terms of the angular rotation of the crankshaft.

In the preferred embodiment, a high-speed hydraulic actuator is positioned to interact with a first (main) camshaft to impart a periodic displacement stroke to the valve via the actuator to open and close the valve according to a cam profile. A second (control) camshaft is positioned to interact with a pressure relief mechanism (valve) on the actuator to discharge hydraulic fluid from within, thereby collapsing the actuator and allowing the valve to return to a closed position regardless of the present orientation of the first camshaft. A full range phasing mechanism is mounted at the driven end of the control camshaft. This allows the control camshaft to be phased in relation to the first camshaft to modulate the width of the control profile and thus valve duration.

In another embodiment a significant increase in the effective valve area is realized by having the main camshaft incorporate "wide" cam lobes. These function to operate the valve during two consecutive strokes, viz. exhaust followed by intake. A further modification is made to the intake system by having a "merged" intake/exhaust manifold wherein the intake and exhaust

passages are joined together to form a straight through passage. Forced air means maintain airflow through the passage so that exhaust gases expelled from the valve orifice are diverted in the discharge direction and fresh air is admitted from the upstream direction. This is a particularly useful feature with variable compression ratio engines wherein a large portion of the combustion chamber surface may be occupied by the sub-piston adversely effecting valve area.

BRIEF DESCRIPTION OF THE DRAWINGS

The above and other embodiments of the present invention may be more fully understood from the following detailed description, taken together with the accompanying drawing, wherein similar reference characters refer to similar elements throughout, and in which:

Figure 1 is a cross sectional view of the arrangement.

Figure 2 is a cross sectional view of another embodiment of the arrangement.

Figure 3 is a cross sectional view of the wide lobe cam.

Figure 4 is a sectional view of the merged intake-exhaust manifold.

Figure 5 is detailed drawing of a high-speed actuator.

DESCRIPTION OF A PREFERRED EMBODIMENT

Referring now to the drawing, Fig 1. shows a sectional view of the arrangement 10 constructed according to the invention as illustrated for one valve 101 in an internal combustion engine.

The valve 101 is shown to be mounted in a partial view of the cylinder head 103. The detail of the valve spring, clips, valve seat etc., are not shown as they are well known in the art and are not taught by the invention.

Lubrication channels 104 which provide a conduit for the engine lubricating oil system are illustrated as formed within the cylinder head 103

A hydraulic actuator 140 is mounted on the cylinder head 103, connected with the lubrication channel 104 and positioned adjacent the valve 101. A camshaft 110 is mounted in the cylinder head above the actuator. A cam 111 is mounted on the camshaft 110. The camshaft will rotate to move the cam lift surface 111 into contact with the actuator top piston 142. The downward movement of the top piston will be transmitted via the pressurized fluid medium 141 to the lower plunger 143 causing it to move valve 101 to the open position. The timing of the valve opening is set by the rotation of the camshaft 110.

The arrangement 10 provides an additional control element, a pressure relief valve 144 formed in the body of the actuator. It is this control element which allows the adjustment of the open duration of the valve 101.

A control camshaft 120 is mounted adjacent the relief valve 144. A control cam with control surface 121 is mounted on the control camshaft and positioned to engage the pin 148 of relief valve 144 for a preselected portion of the rotation of the control camshaft.

The control camshaft may be driven by an arrangement taught in U.S. Patent 4,747,375, ('375) granted to John K. Williams. The '375 arrangement allows a crankshaft to drive a camshaft with variable phasing over the entire load range. The present invention utilizes this variable phasing unit to adjust the engagement of the control surface 121 with the pin 148 of the relief valve 144.

One advantage of this system is that the control camshaft can be made much lighter and at less cost than the main camshaft since it interacts only with the pressure relief mechanism. Accordingly the phasing unit is also very light allowing it to be directly connected to an accelerator pedal.

Figure 2, shows another embodiment of the present invention wherein the actuator 140 interacts with the valve 101 via a rocker arm. A rocker arm 102 is hingedly mounted to the top portion of the actuator. A camshaft 110 is mounted over a plurality of rocker arms (one for each valve). A cam 112 is mounted on camshaft 110 and is in sliding contact with rocker 102. The cam 112 shown in this case is the wide lobed cam mentioned above.

At the start of a sequence, camshaft 110 will rotate to move the cam lift-surface 112 into contact with the rocker arm 102. The rocker arm 102 will pivot from the hinged mounting to begin to push the valve 101 to an open position. The timing of this event coincides with the beginning of the exhaust stroke of the

particular cylinder. The valve remains open during the exhaust stroke and into the intake stroke.

The control camshaft 120 is mounted adjacent actuator 140, where control cam surface 121 is engageable by relief pin 148 as before. When upon further rotation of control cam 121, the control surface causes the relief pin to be pushed inward to release the hydraulic fluid the actuator 140 will contract to move pivot point 140 downwards as viewed in the drawing. This causes a corresponding upward movement of the opposite end of rocker 102 to bring valve 101 to a closed position under the influence of the valve closing spring. The top surface of the rocker 102 is so shaped as to facilitate the rocker to pivot about a second pivoting point formed by the sliding contact point of cam 112 and rocker 102.

A further advantage of this embodiment is that actuator wear is reduced and the actuator can be made simpler since it executes a single function stroke per cycle rather than the double stroke of the first embodiment. Thus in this mode the actuator can function with only a single movable plunger.

It should be noted that the wide cam lobe 112 may also be employed in the first embodiment shown in Fig. 1. If needed, the lobe 112 may incorporate a slight depression or valley approximately in the middle to enable the valve to retract slightly while the piston 160 is near the top of its stroke.

Figure 4 shows an embodiment of the merged intake-exhaust system employed in conjunction with a dual function valve. The manifold 150 comprises an upstream portion 151 and a downstream portion 152 respectively to valve 101. An integral roots or scroll type blower 153 driven by the crankshaft is shown attached to the manifold forcing air through conduits 151 and 152. Upon opening of the valve 101 the high pressure exhaust gasses are vented to the passage and forced in the discharge direction by the airstream generated by the blower. This process continues while the piston 160 moves up to the top dead center (TDC). As soon as pressure equilibrium is reached fresh air starts to fill the cylinder space under the influence of the airstream. On the subsequent intake stroke the piston 160 begins to move down evacuating more space and drawing in more air. At some point during this stroke valve 101 is closed under the control of the valve control system, shutting of further supply of charge to the cylinder.

The fuel injection pulse begins approximately when the valve is near TDC (soon after the gas equilibrium point is reached), and stops before the valve 101 is closed. The fuel injection pulse width may be synchronized with the valve

duration pulse, or injection may be through a separate valve port in the cylinder operated in the conventional manner.

A separate smaller valve may be positioned across from the main valve(s) in the combustion chamber to open momentarily near the TDC point to flush out trapped gasses. In the preferred embodiment the passage 150 is shown to turn sharply at an acute angle at the point of communication with the cylinder. This creates a ramming effect by the airstream facilitating cylinder charging at high revolutions.

Figure 5 shows the detailed design of a high speed actuator. As mentioned earlier, most actuator designs are plagued by slow response times which result in serious degradation of the of the operational profile at high rpm. Many designs, to enhance response time, store the expelled hydraulic fluid in a pressure reservoir. Since the reservoir is connected via conduit, delay arises due to path restriction. Also, transfer of fluid between actuator and reservoir is effected by solenoids, adding to the cost of the units.

The actuator shown in Fig. 5 has an integral pressure reservoir to store the expelled fluid and return it in the shortest possible time during the neutral interval. In the preferred embodiment, the actuator 140 encloses an internal volume space 141 formed by the casing and movable plungers 142 and 143. Plungers 142 and 143 are formed with a second smaller diameter to slidably fit into a smaller cylindrical bore formed within the body of the actuator. This reduces the volume of fluid displaced for a proportional relative movement of the plungers improving response time. The first volume chamber 141 communicates with a second volume chamber 145 by way of a check valve 144. Check valve 144 has a large bore diameter respective to the volume chamber to facilitate rapid transfer of fluid. The second volume chamber 145 has a movable piston 146 biased by spring 147 in the direction of the check valve 144. A pin 148 slidably passes through a bore in the actuator body and a second bore in the slidable piston 146 to communicate with the check valve 144. Inward motion of the pin 148 opens check valve 144 allowing hydraulic fluid to enter volume 145 forcing piston 146 against biasing spring 147. The pin is flanged at two points to limit travel. Upon completion of a stroke, when pressure of the cam face on plunger 142 is relieved, fluid from chamber 145 is forced back into chamber 141 by the energy stored in spring 147. Piston 146 and 142 are shown hollowed out in the middle for compactness and low mass.

Another check valve 149 admits hydraulic fluid from the engine lubrication system to prime the actuator initially. Another feature of this design is that the spring coefficient 147 can be matched to the coefficient of valve spring 105 to determine the valve return rate in order to avoid excessively high valve seating velocities. Alternately, damping can also be achieved by having fluid occupy the back space of piston 146 and by spilling this fluid through calibrated spill bores.

In the preferred embodiment, chamber 145 incorporates a small bore at a preselected volume to bleed off excess fluid. The base of the actuator is flanged for mounting on the cylinderhead. An oil inlet passage is shown which mates with a corresponding oil supply channel 104 from the engine lubrication system. Another spring is mounted between the outer peripheral flange of plunger 142 and housing 140 to urge the plunger against the cam face and improve response.

In an alternate embodiment relief pin 148 may be operated by small two-position solenoids attached to the actuator. Signals from an ECU can then be employed to operate the actuators. In some modes of operation a number of valves in each cylinder may be disabled to increase the flow velocity into the cylinders. This can also be done by having split lobes for the control cam 121 and engaging pin 148 before intake begins for some of the valves.

In yet another mode of operation the actuators may be used in conjunction with a conventional throttle valve in which the actuators are altogether disabled (in the fully extended position) at high rpm and charge flow is controlled in the usual manner for the upper load ranges.

Accordingly, the reader will see that the above invention demonstrates a vastly improved and efficient valve control system for an engine's valves. Furthermore, the system overcomes many of the disadvantages of the prior art in a very cost effective manner. System reliability is improved, while at the same time energy drain and stress on moving components is reduced.

Since certain change may be made in the above apparatus without departing from the spirit and scope of the invention herein involved, it is intended that all matter contained in the above description, as shown in the accompanying drawing , shall be interpreted in an illustrative and not a limiting sense.

I claim:

1. A system for controlling fluid flow to and from the valves of an internal combustion engine, comprising:

a cylinder having a valve, an inlet passage connected to deliver air to said valve;

defining an outlet passage to receive exhaust gas from said valve, said inlet passage and said outlet passage being connected together adjacent said valve;

means for providing gas flow through said inlet passage and said outlet passage.

2. The passage of claim 1 further characterized in that said inlet passage being formed as separate conduits each communicating with its respective valve;

said conduits merged into a common intake manifold upstream of said valve; said outlet passage being formed as separate conduits and merged into a common discharge passage downstream of said valve.

3. The arrangement as claimed in claim 2 wherein reed valves are disposed in said conduits either downstream or upstream of said valve to prevent backflow of fluid.

4. A variable timing system for an internal combustion engine with reciprocating pistons comprising:

a valve controlled by a cam on a camshaft via the interposition of a corresponding actuator means between said valve and said cam;said cam having a base circle, a lift surface, ramp areas joining the base circle and lift surface;

said lift surface having an angular range corresponding rotationally at least to a greater portion of two consecutive strokes of said piston;

said actuator means adjustable in length while in operation whereby the radial displacement produced by the throw of said cam lift surface is reversed at any intermediate angular point of said lift surface.

5. The arrangement described in claim 4 wherein said actuator has means to adjust its effective length by the selective pumping of hydraulic fluid into and out of an expansible and contractible interior hydraulic chamber space.

6. The arrangement described in claim 5 wherein said actuator is interposed between said cam and a hinged rocker positioned to interact with said valve.

7. A hydraulic engine valve timing system for individually controlling the operation times of the cylinder valves;

the system having a first rotational camshaft means having a first cam means respective to each said valve, a hydraulic actuator means interposed between said valve and said first cam means to transmit reciprocating motion to the respective valve;

said actuator means adjustable in effective length to overcome stroke of first cam means by the release of hydraulic fluid from a contractible interior hydraulic volume space ;

a second rotational control camshaft means having a control cam means adjacent each said actuator means; said control cam means interactive with said actuator means to periodically discharge hydraulic fluid from said actuator means and thereby effect a reversal of axial displacement produced by said first cam means.

8. The invention of claim 7 further comprising:

shaft phasing means mounted on said control camshaft means to phase said control camshaft means with respect to said first camshaft means.

9. The invention of claim 8 further comprising a shaft phasing means mounted on said first camshaft means.

10. A hydraulic valve actuator having a housing, a first variable volume chamber enclosed by said housing and at least one movable surface slidably disposed in said housing positioned to vary the effective height of said actuator;

a second volume chamber enclosed in a second portion of said housing and communicating with said first volume chamber through the imposition of a checkvalve therebetween;

said second volume chamber having a slidable plunger means disposed therein, spring means biasing said plunger means in a direction of lower volume;

means provided to operate said check valve means to pass fluid from said first volume chamber to said second volume chamber under a condition of fluid pressure in said first volume space being greater than fluid pressure in said second volume space.

11. The arrangement of claim 10 further comprising: a second check valve means disposed in said first volume space and connected with external fluid supply means to admit fluid to said first volume space under a condition of fluid pressure in said first volume space being lower than fluid pressure of said external supply means.

12. The arrangement of claim 11, further comprising a small bore disposed at a preselected volume of said second volume space.

13. The arrangement of claim 10, wherein flow capacity of said checkvalve means is large in relation to said first volume space.

14. The arrangement of claim 10, further comprising linkage means to engage rotating cam surfaces thereby to effect opening of said checkvalve means.

15. The arrangement of claim 10, further comprising solenoid switch means to operate said checkvalve means.

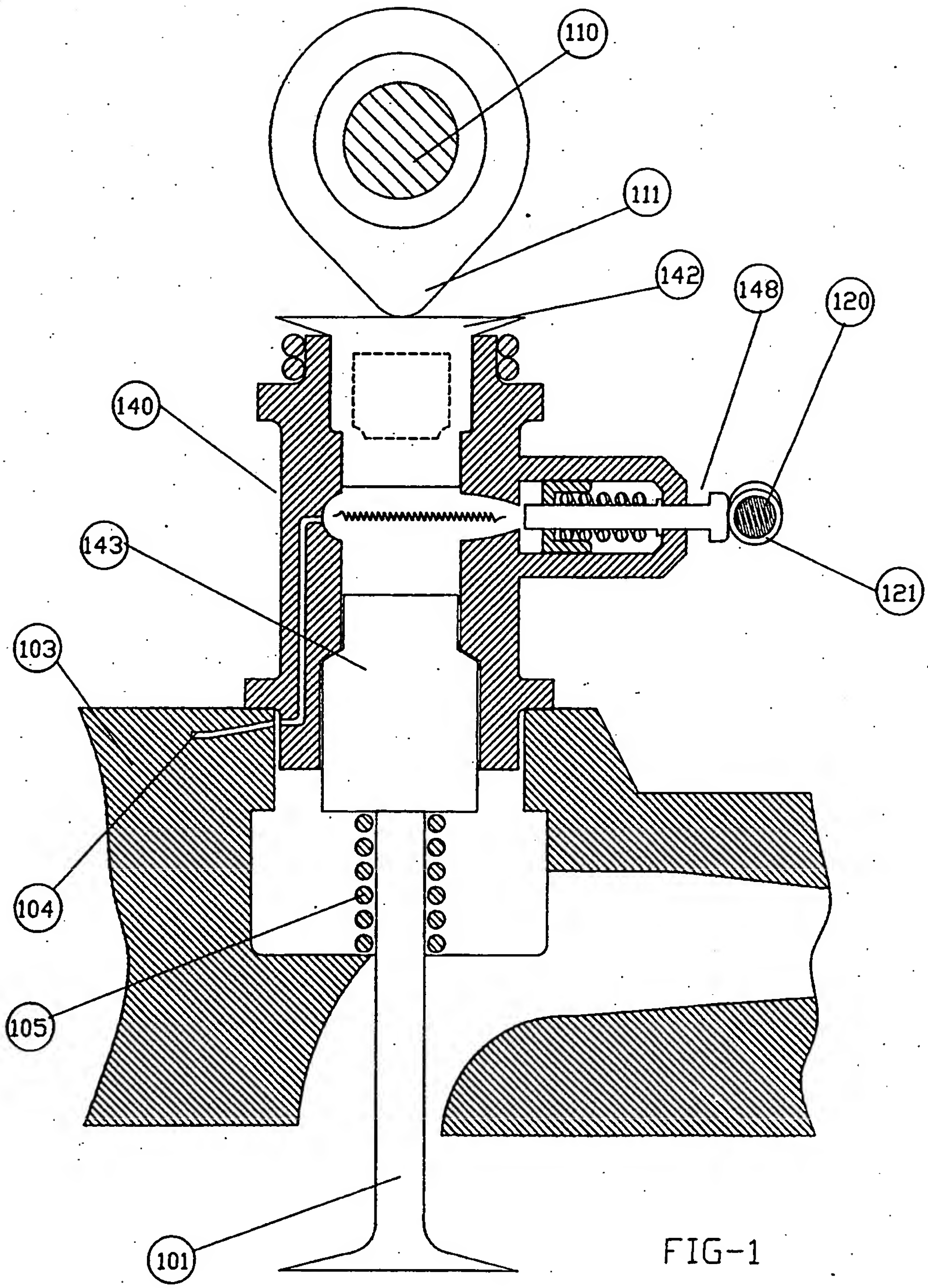

FIG-1

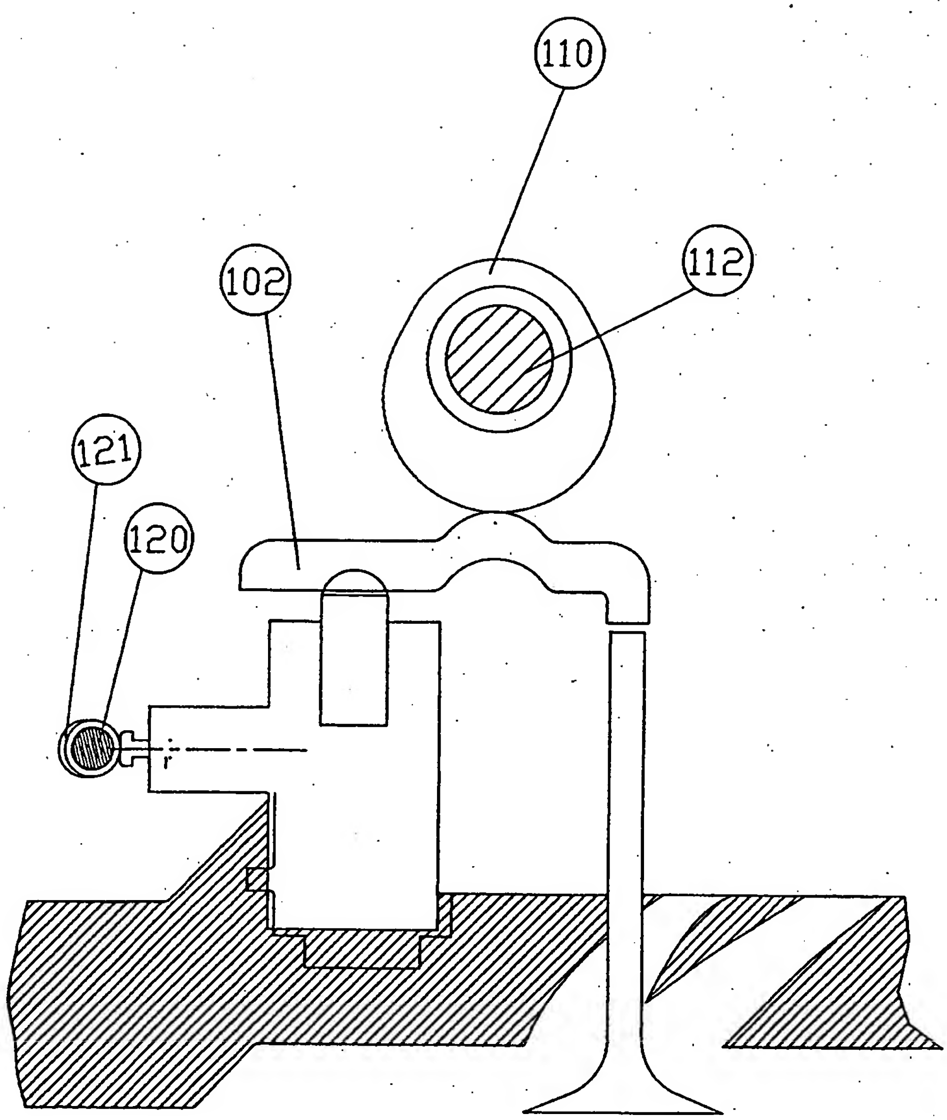

FIG-2

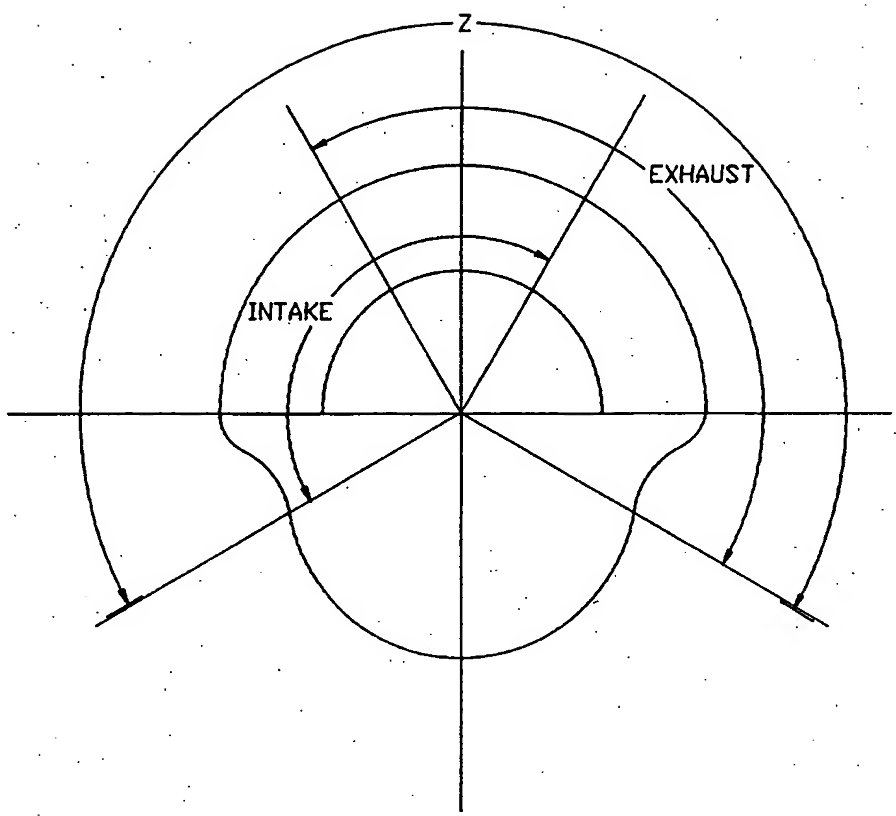

FIG.3

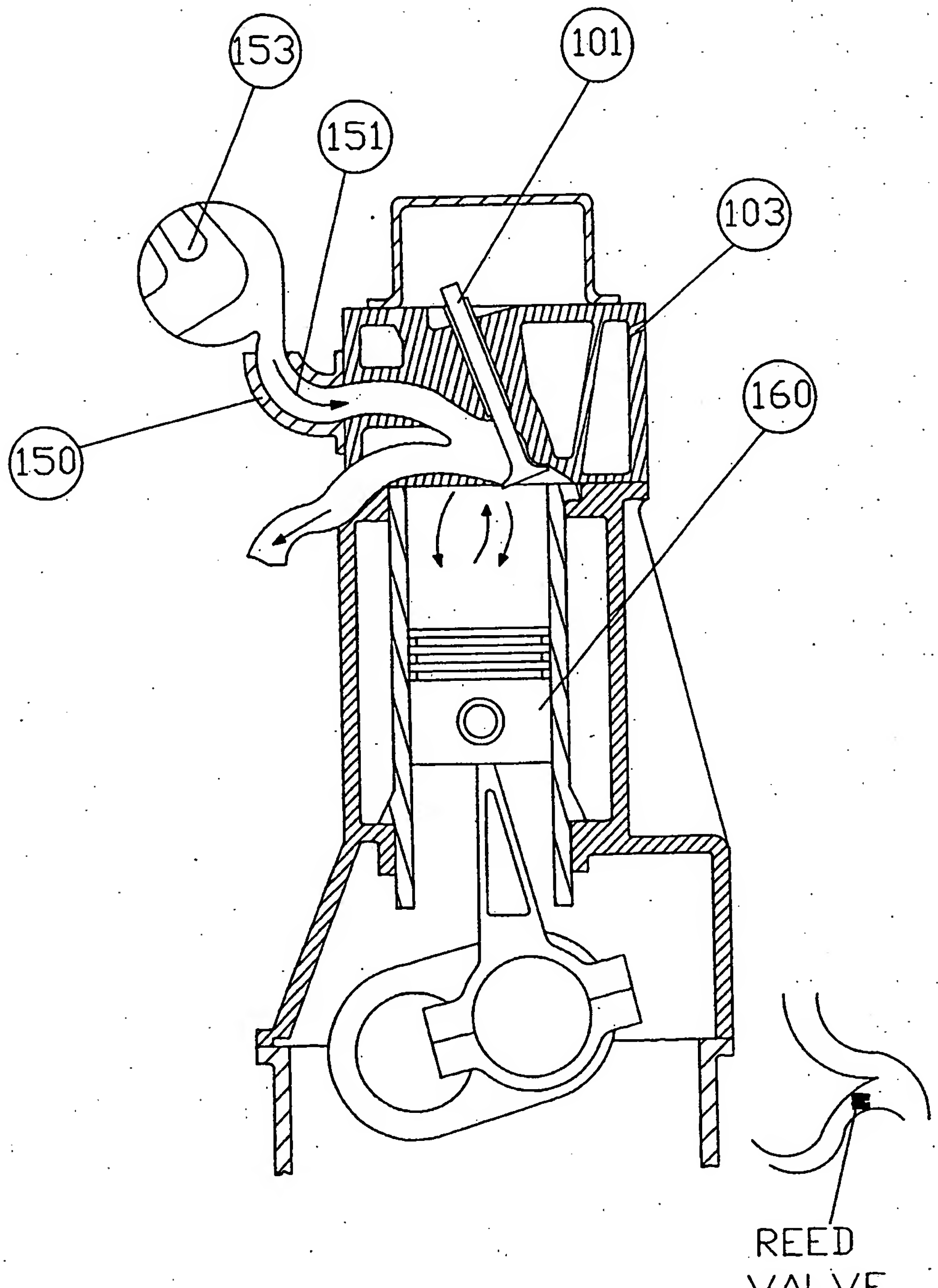

FIG-4

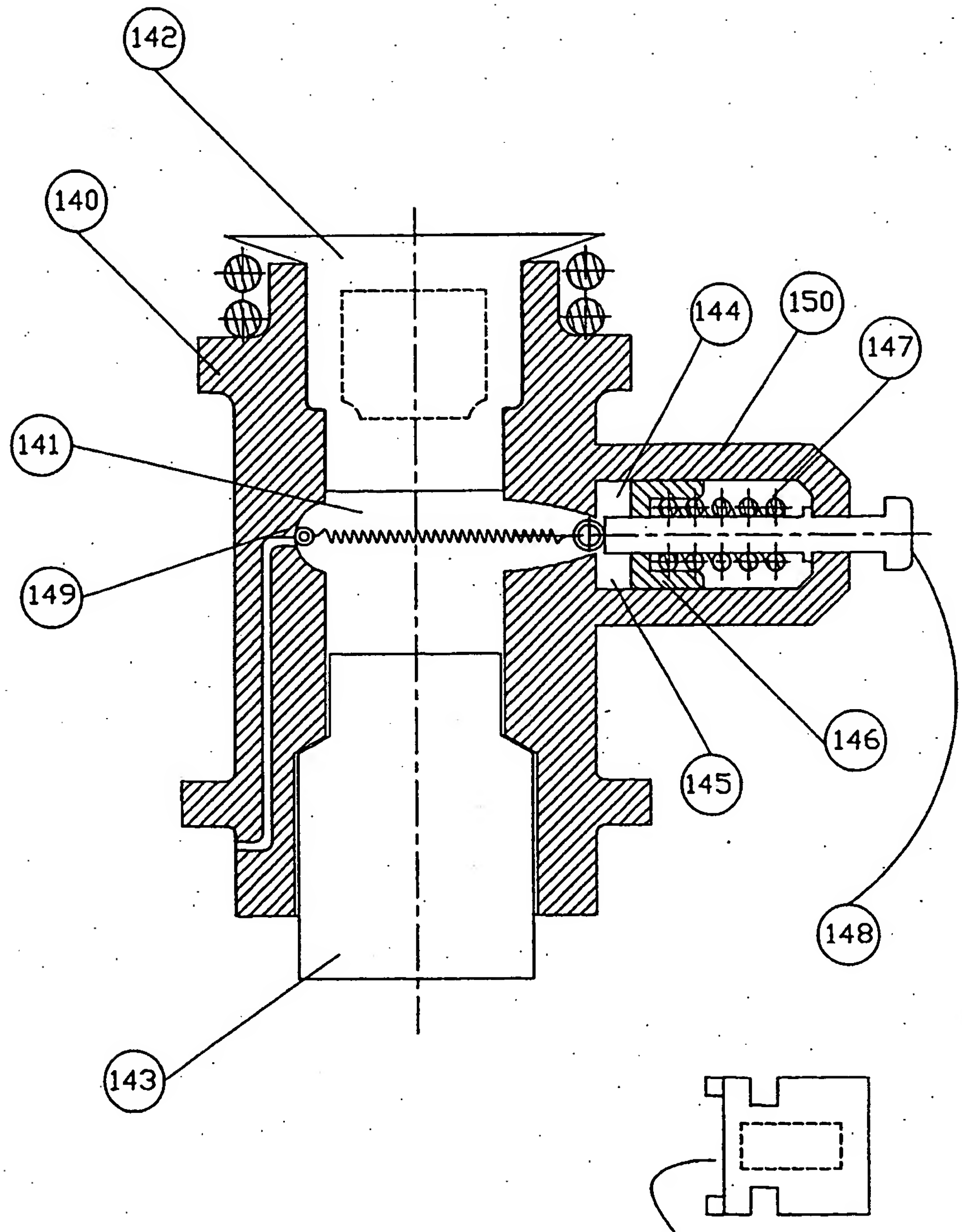

FIG-5